NACA RM-E51A03

RESTRICTED

Copy 48 c.1
RM E51A03

NACA

RESEARCH MEMORANDUM

A SUMMARY OF DESIGN INFORMATION FOR WATER-COOLED TURBINES

By John C. Freche

Lewis Flight Propulsion Laboratory
Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

WASHINGTON
March 9, 1951

RESTRICTED

A topic outline of each factor that must be considered to arrive at an acceptable water-cooled-turbine design is presented. Each topic is discussed in an ensuing section. The outline is subdivided into two major sections. The first section presents the underlying principles and calculation methods required to make and to evaluate a water-cooled-blade design. The second section provides information on various material, fabrication, and engineering problems, which should be considered in a water-cooled-turbine design.

WATER-COOLED-TURBINE DESIGN OUTLINE

I. DESIGN PRINCIPLES AND CALCULATION METHODS

A. Principles of Coolant Circulation
  1. Forced convection
  2. Natural convection
  3. Combination (cross-over holes)
  4. Pumping by free convection

B. Positioning of Coolant Passages
  1. Location of maximum blade temperature
    a. Experimental blade-temperature distributions
    b. Limitations due to stress
      (1) Minimum wall thickness
    c. Elliptical coolant passages
    d. Coolant passages $<$0.060-inch diameter
      (1) Fabrication difficulties
      (2) Friction pressure drop
    e. High-conductivity-material inserts

C. Determination of Blade-Design Suitability
  1. Allowable blade-temperature distributions
  2. Actual blade-temperature distributions
    a. Experimental gas-to-blade heat-transfer coefficients
    b. Method of computing gas-to-blade coefficients
      (1) Stream-filament theory
    c. Experimental blade-to-coolant coefficients required
    d. Spanwise blade-temperature distribution
    e. Chordwise temperature distribution at trailing edge
    f. Chordwise temperature distribution at leading edge
      (1) Variable conductivity

II. ENGINEERING CONSIDERATIONS

A. Selection of Blade Material
   1. Availability
      a. Strategic materials
      b. Nonstrategic materials
   2. Strength
      a. Stress-to-rupture properties
   3. Thermal conductivity
      a. Variation with temperature
   4. Fabrication
      a. Machinability
      b. Casting properties

B. Cooled-Blade Manufacture
   1. Drilling
   2. Casting
   3. Disintegration method
   4. Fabrication of high-conductivity inserts

C. Disk Design
   1. Constant-strength-disk method
   2. Methods including effect of thermal gradients

D. Coolant Sealing
   1. Sealing at points of entrance and exit from rotor
   2. Sealing at blade tips
   3. Sealing between blades and disk
      a. Blades not integral with disk
      b. Compromise between integral and individual blade construction

E. Stator-Blade Cooling
   1. Coolant passages
      a. Equal flow to all coolant holes
   2. Cooled annulus

F. Water Purification
   1. Material deposition in passages
   2. Corrosion
   3. Purification systems
      a. Chemical
      b. Distillation

NATIONAL ADVISORY COMMITTEE FOR AERONAUTICS

RESEARCH MEMORANDUM

A SUMMARY OF DESIGN INFORMATION FOR WATER-COOLED TURBINES

By John C. Freche

SUMMARY

A tabulation of information specifically applicable to the design of water-cooled turbines is presented in this report. A design procedure has been set up. Required calculation procedures described in other publications are referenced. Supplementary discussions of pertinent data that do not appear in published sources are provided.

INTRODUCTION

Substantial increases in the power output of both turbojet and turbine-propeller power plants can be obtained with high turbine-inlet gas temperatures. Turbine-inlet gas temperatures are limited, however, by stress limitations in current high-temperature materials. One method of achieving higher inlet gas temperatures is by cooling. Water cooling, particularly, promises large increases in turbine operating temperatures.

Theoretical analyses of various phases of turbine cooling are being carried on at the NACA Lewis laboratory. Simultaneously, experimental investigations are being made with two water-cooled turbines, one utilizing the principle of forced-convection circulation, and the other the principle of natural-convection circulation. These turbines were designed and built at the NACA. As a result of theoretical analyses and experimental investigations, much information pertaining to the design, construction, operation, and cooling effectiveness of water-cooled turbines has been accumulated.

In order to facilitate the use of this information by prospective designers of water-cooled turbines, an attempt has been herein made to tabulate and where possible make reference to all material specifically applicable to the design of water-cooled

turbines. An outline of design procedure is presented to assist designers in obtaining a clearer perspective of the over-all problem. Supplementary discussions of unpublished information, which could not be referenced, are provided. This report provides a digest of information currently available, which may serve as a guide to prospective builders of water-cooled turbines, but it should not be considered as a comprehensive design manual.

GENERAL CONSIDERATIONS

In order to design a water-cooled turbine, a tentative aerodynamic design should first be established using conventional methods with consideration for limitations that will subsequently be discussed. For an existing uncooled turbine, such tentative aerodynamic modifications as are necessary to achieve the design objective should be incorporated. Once a tentative aerodynamic blade design is selected, the various cooling and circulation methods should be considered. These are forced convection, natural convection, or a combination of both. Selection of the circulation system will eventually determine the following:

(a) Size, number, and location of blade coolant passages

(b) The need and extent of special measures for control of leading- and trailing-edge temperatures

(c) The general configuration of the turbine rotor and coolant-supply system.

In order to select the circulation system most favorable to a given design objective, an analysis should be made of the coolant-flow characteristics considered as functions of the operating conditions imposed on the turbine and the physical characteristics of the turbine material. Various factors contributing to the final design configuration are interrelated. For example, determination of the cooling characteristics is dependent upon (1) the gas flow and temperature level which have been set as the design objective and (2) the blade material. Blade material affects blade-temperature distribution because the distribution is greatly dependent upon thermal conductivity. Also blade material limits the maximum blade temperature for the conditions of centrifugal, bending, and thermal stresses encountered. Furthermore, the selection of the blade material is affected by strategic considerations and construction considerations particularly as applied to coolant-passage and rotor fabrication.

TOPIC OUTLINE DISCUSSION

Design Principles and Calculation Methods

I-A-1. Forced convection

The principle of forced-convection circulation of water through turbine blades is presented in reference 1, which deals with an experimental investigation of a small, forced-convection, water-cooled turbine.

I-A-2. Natural convection

The principle of natural-convection circulation through turbine blades is explained in references 1 and 2. Reference 1 also deals with an experimental investigation of a natural-convection, water-cooled turbine.

I-A-3. Combination forced and natural convection (cross-over holes)

The principle of augmenting natural-convection circulation by utilizing a large supply hole that connects with the smaller-diameter holes is described in reference 2. The increased effectiveness obtainable over simple natural convection is indicated by calculation in reference 2.

Lack of sufficient experimental data prevents drawing a final conclusion as to which method of cooling is best. A recommendation can be made, however, on the basis of data available. Comparison of the blade-to-coolant heat-transfer coefficients obtained at similar gas state conditions indicates the forced-convection turbine has coefficients about three times as high as the natural-convection turbine. It should be emphasized that higher coefficients than were obtained with the NACA natural-convection turbine might result from natural-convection turbines having larger coolant-passage diameters. If greater coolant flows are utilized in the forced-convection turbine, the coefficients will increase still more due to the greater turbulence at higher Reynolds numbers. At present, the flow to the forced-convection turbine is limited by the coolant supply pressure. Use of a pumping system such as the one described in section I-A-4 would provide a tremendous pressure for circulating the coolant, thus achieving the higher Reynolds numbers and consequently higher coefficients. On the basis of present knowledge,

therefore, forced-convection cooling with the pumping force provided by free convection appears the most promising.

I-A-4. Pumping by free convection

An important factor in a forced-convection circulation system through a liquid-cooled turbine is that a large pressure difference is created between the points of coolant entrance and exit from the rotor in a continuous-flow system. Actually a free-convection pumping force is initiated through the turbine. Referring to figure 1, the density of the coolant flowing into the rotor (passage A) is greater than that leaving the rotor (passage B), because heat transfer has increased the coolant temperature. Thus the turbine acts as a pump setting up a pressure differential between the coolant entering and leaving the rotor. A formula expressing this pumping force can be written as

$$\text{PUMPING FORCE} = \left[\frac{(R_2^{\ 2}-R_1^{\ 2})\omega^2}{2g}\right](\rho_A-\rho_B)$$

where

R_1 radius to point of coolant discharge, (ft)

R_2 radius to top of coolant passages, (ft)

ω turbine speed, (radians/sec)

g acceleration of gravity, (ft/sec^2)

ρ_A average density of coolant in inlet passage, (lb/cu ft)

ρ_B average density of coolant in exit passages, (lb/cu ft)

This pumping force may be used to pump the coolant through the external system of filters and heat exchangers as well as through the turbine, thereby eliminating the need for an auxiliary high-capacity pump. The size of booster pump required to initiate flow and provide forced circulation until sufficiently high turbine speeds are achieved, as well as the type of valving arrangements required, are design problems and dependent upon individual turbine applications. For a turbine with 100 blades of 2-inch span, having nine 0.020-inch coolant passages per blade and assuming a total coolant flow of 9 gallons per minute

through the rotor and a specific coolant temperature distribution, a pumping force of approximately 325 pounds per square inch was calculated.

It is interesting to consider a modification of this method. In reference to figure 1, assume that the water inlet and outlet are connected at the center of the shaft, thereby providing circulation through a continuous or loop circuit. A valve to regulate the flow would be placed in this connecting line. A recirculation of the coolant through the loop circuit would result in raising the coolant-temperature level as well as decrease the amount of coolant used. Consequently, a more effective use of the coolant can be achieved. The principle of pumping by free convection is still maintained so long as sufficient cold water is introduced into the circuit through the water inlet to maintain a temperature differential between columns A and B. The interconnection of the water inlet and exit, however, provides a useful design modification.

I-B-1-(a,b,c). Positioning of coolant passages - location of maximum blade temperatures

Theoretical analysis has indicated that maximum temperatures in cooled blades will occur at the leading and trailing edges. Experimental blade-temperature distributions (reference 1) have indicated that maximum temperatures in water-cooled turbine blades occur at the leading and trailing edges. Additional turbine operation has continued to substantiate these findings. The effective length of the uncooled portion of the blade leading and trailing edge must therefore be kept to a minimum in order to achieve sufficient cooling at these locations. Calculations were made for the turbine shown in figure 2 by means of analytical equations and experimental heat-transfer coefficients (see section I-C-2) and these indicated that a maximum trailing-edge length of 0.250 inch could be tolerated. However, moving the cooling passage close to the trailing edge necessitates decreasing the passage diameter so that it is less than that of the other coolant passages in the thicker portion of the blade. This may result in starving the flow to the small coolant passage and cause ineffective cooling. Also the problem of minimum wall thickness must be considered. For a given trailing-edge configuration, the wall thickness is successively reduced as a fixed-diameter circular passage is moved closer to the trailing edge. Excessive water pressures due to high rotational speeds and the necessity for allowing sufficient material to accommodate drill drift if the blade passages are drilled require a certain wall thickness. Experiment has indicated that 0.030 inch is satisfactory as a wall thickness for the 14-inch turbine-blade configuration (reference 1). The use of an elongated coolant passage of the same cross-sectional area as a circular passage would serve to decrease the trailing-edge length; however, the tendency of such a passage configuration to assume a circular shape upon rotation may result in excessive stresses. Use of such a passage configuration is therefore not recommended for rotor blades; however, its use in stator blades is feasible. (See section II-E.)

I-B-1-d. Coolant passages less than 0.060-inch diameter

The smallest-diameter coolant passage on current NACA experimental water-cooled turbine blades is 0.060 inch. Stress and fabrication difficulties, which determine the minimum wall thickness, prevent bringing coolant passages sufficiently near to blade leading and trailing edges to achieve a high blade-to-coolant heat rejection in these sections of the blade. By reducing the passage diameter below 0.060 inch, many more coolant passages may be placed in a blade, and coolant passages may be located nearer to the leading and trailing edges of the blades as well. Also, for constant coolant weight flow, increased turbulence in the small-diameter coolant passages results in higher blade-to-coolant heat-transfer coefficients in these passages. Thus, for a given coolant flow through the turbine rotor, use of small-diameter coolant passages can provide better cooling by the more advantageous positioning of coolant passages as well as by the higher individual heat-transfer coefficients that result. Two major factors may hinder use of coolant-passage diameters less than 0.060 inch, however. Specifically, these are fabrication difficulties and possible excessive friction pressure drops, which may result in high pumping losses. Fabrication considerations are discussed in section II-B-1,2,3. The actual friction losses in small holes must still be determined experimentally. However, using friction factors obtained from pipe diameters 3/8 inch and larger, preliminary calculations have been made which indicate that friction pressure losses may not be excessive. An idea of the magnitude of these losses can be obtained from the following example. For a turbine with 100 blades of 2-inch span, having nine 0.020-inch coolant passages per blade, the friction pressure drop in the blades was calculated to be 4.75 pounds per square inch (9.6 in. Hg), assuming a total coolant flow of 9 gallons per minute through the turbine and a specific coolant temperature distribution. A better indication of the effective magnitude of this friction loss can be obtained by comparison with the total (free-convection) pumping force available for a turbine. (See section I-A-4.) The blade friction pressure drop is approximately $1\frac{1}{2}$ percent of this total pumping force.

I-B-1-e. High-conductivity-material inserts

In cases where the coolant passage cannot be moved sufficiently close to the trailing edge because of aerodynamic or mechanical design considerations, reduction of trailing-edge temperature may be achieved in another manner. The thermal conductivity of the trailing-edge blade material may be effectively increased, thereby facilitating heat transfer from the blade trailing edge to the coolant. This principle has already been successfully applied with air-cooled

blades in which a copper strip was brazed into a slot cut into the blade trailing edge. The method of fabrication of these inserts is discussed in section II-B-4. Two static-wedge designs that duplicate the trailing-edge section of the 14-inch liquid-cooled turbine (reference 1) and that have copper inserts are being considered. The insert extends from the trailing edge to the water passage on one wedge and from the trailing edge to within 1/32 inch of the water passage on the other. If no appreciable temperature difference exists between the two configurations, the configuration in which the insert does not contact the coolant passage should be used in a turbine installation because the braze is then not subject to the additional stresses of water under high pressures.

I-C-1. Allowable blade-temperature distributions

In order to obtain the safe operating temperature limit for cooled turbine blades, an allowable radial-temperature-distribution curve should be determined. The allowable blade temperature naturally depends upon the stress characteristics of the blade. The chief types of stress concerned are centrifugal, vibratory, and thermal. The effect of the last two types are extremely difficult to evaluate and all these stresses are superimposed upon each other. The current method of obtaining the allowable-temperature-distribution curve is one dimensional and is based upon the radial-centrifugal stress distribution in the blade, as well as stress-to-rupture data for the blade material being used. This method is fully described in reference 3.

I-C-2-a. Actual blade-temperature distribution - experimental gas-to-blade heat-transfer coefficients

Having obtained the allowable blade-temperature distributions, an actual blade-temperature distribution for a desired set of operating conditions (design conditions) must be determined to see if the design conditions can be safely reached. Theoretical equations have been derived by means of which blade temperatures can be calculated. However, to calculate these blade temperatures, gas-to-blade and blade-to-coolant heat-transfer coefficients are required.

Gas-to-blade heat-transfer coefficients can be obtained from the published experimental results of other investigators of gas-to-blade heat transfer to various blade shapes in static cascades. These results are compiled in reference 4. Recently, experimental gas-to-blade coefficients were obtained on a small water-cooled turbine. Comparison of these data with experimental results from a static

cascade of similar shaped blades showed agreement within 9 percent (reference 5). Such agreement gives additional credence to the assumption that static-cascade results may be used to approximate turbine gas-to-blade heat-transfer rates.

I-C-2-b. Method of computing gas-to-blade coefficients

Frequently, experimental heat-transfer results obtained on a static cascade are not available for a blade configuration similar to the turbine-blade design contemplated. A theoretical method has been developed by the NACA (reference 6), which employs boundary-layer equations for computing the gas-to-blade heat-transfer correlation for turbine blades. In order to utilize the method set up in reference 6, the blade peripheral velocity distribution must be known. The velocity distribution may be determined by stream-filament theory (reference 7). The usefulness of the method of computing gas-to-blade heat transfer is pointed out in reference 6 where agreement within 15 percent with experimental results from static-cascade investigations is illustrated. Further verification of the method is provided in reference 5. In this reference, agreement within 3 percent is shown between cooled-turbine gas-to-blade heat-transfer data and results computed by the theory of reference 6 using a blade velocity distribution calculated by stream-filament theory.

I-C-2-c. Experimental blade-to-coolant coefficients required

Blade-to-coolant heat-transfer coefficients may be obtained from static-tube data when a forced-convection method of coolant circulation is being used. Complete verification of the use of such data for application to cooled turbines has not yet been obtained. However, blade-to-coolant heat-transfer data have been correlated for a forced-convection water-cooled turbine. Correlation was achieved on the basis of Graetz number (which is primarily a function of the flow rate), and the turbine data were displaced above the static-tube data. It is suggested that static-tube data correlated in this manner (reference 8) be used to obtain blade-to-coolant heat-transfer coefficients for use in computing the actual blade-temperature distribution. Blade-to-coolant heat-transfer data have also been correlated for a natural-convection water-cooled turbine. Correlation was achieved on the basis of Grashof number (primarily a function of the gravitational or accelerating forces acting on the coolant and the temperature differential causing heat transfer), although considerable scatter resulted (reference 1). It is suggested that this correlation be used to obtain blade-to-coolant heat-transfer coefficients for natural-convection turbines having passage configurations similar to the 14-inch water-cooled turbine described in reference 1. Reference 2

should be consulted for natural-convection turbines with larger coolant-passage diameters.

I-C-2-(d,e,f). Actual blade-temperature distributions

Analytical methods of computing the spanwise and chordwise blade temperature distributions are derived in references 9 to 11.

A one-dimensional method utilizing equations derived in reference 9 may be used in obtaining a spanwise temperature distribution near the coolant passage. When a temperature distribution remote from the coolant passage is desired, it is recommended that equation (37), reference 10, be used. This equation was obtained by approximating the section with a rectangular parallelepiped and heat flow in three directions was considered.

Other temperature distributions throughout the blade can be calculated in the following manner. A chordwise temperature distribution along a mean camber line in the trailing-edge section of a blade can be obtained by approximating the trailing-edge section with a rectangle when rim-cooling effects are not appreciable (equation (11), reference 10). More accurate results can be obtained by approximating the trailing-edge section with a trapezoid and using equation (25), reference 10.

The methods mentioned apply to the blade leading-edge section as well, if the blade configuration is such that it may be approximated by a trapezoid or rectangle. Frequently, the annular section formed by concentric circles approximates the leading-edge section of a blade. In that case use of equations that are derived in appendix B, reference 11, will give a more accurate temperature distribution from the coolant passage to the blade leading edge.

Use of a high-conductivity insert in the blade trailing edge (section I-B-1-e) presents the problem of how to account for the difference in conductivity of the blade and insert material in calculating trailing-edge-blade temperatures. A possible method is to consider that a metal coating (the blade metal) is interposed between the insert and the hot gas. An over-all coefficient U from hot gas to metal can then be written as

$$\frac{1}{U} = \frac{1}{q_i} + \frac{t_c}{k_c}$$

where

q_i heat-transfer coefficient from hot gas to metal

t_c thickness of coating

k_c conductivity of coating

The over-all coefficient U then replaces q_i in determining the value C, which is used in equation (25), reference 10.

ENGINEERING CONSIDERATIONS

II-A-1. Selection of blade material - availability

In time of national emergency, the shortage of various critical materials found in heat-resistant alloys makes the selection of turbine-blade material of great importance. From the standpoint of availability, therefore, such heat-resistant alloys as Inconel X, X-40, and AMS 5385, to mention a few, should be avoided and alloys of a low strategic-material content should be selected. The necessity for cooling becomes apparent so that alloys of low strategic-material content can be operated at temperatures sufficiently low to stay within the permissible strength range and still permit high-inlet gas temperatures for high turbine output. An important factor common to some alloys of low strategic-material content is that they are actually better strengthwise than the heat-resistant alloys up to temperatures of 800° to 900° F. Their strength decreases rapidly beyond this temperature, whereas the heat-resistant alloy maintains a virtually constant strength value up to approximately 1200° or 1300° F. The net gain occurs between 800° and 1300° F. If a turbine can be sufficiently cooled, therefore, the use of noncritical material alloys is actually desirable from a strength standpoint. The order of importance of the critical materials as determined by the military is a constantly changing one and naturally constitutes confidential information. However, to serve as a guide, the six most critical materials, not necessarily in the order of their importance, are cobalt, columbium, chromium, nickel, tungsten, and molybdenum.

II-A-2. Selection of blade material - strength

The strength properties of various materials can be obtained from many sources. A source of particular value is the trade literature of the manufacturers of various alloys. Obviously, a complete list of strength properties of the various materials under consideration

cannot be presented here. A table of the 1000-hour stress-to-rupture life of several common turbine-blade materials is included, however, to show the order of magnitude that is involved.

Alloy	Stress-rupture strength (lb/sq in.)	
	Temperature, $^{\circ}F$	
	1350	1500
S816	29,000	21,500
Haystelloy-B	25,000	11,000
Nimonic 80	23,000	12,000
Inconel X	40,000	18,500

References 12 to 14 are good sources from which strength properties of a great variety of materials over wide temperature ranges can be obtained.

II-A-3. Selection of blade material - thermal conductivity

Theory and experiment indicate the importance of the thermal-conductivity properties of a material for application to turbine blades. The higher the conductivity of a material the better cooling may be effected, particularly where the heat flow path is long, as in blade trailing edges. A material cannot be chosen for its thermal-conductivity properties alone, however. A compromise between various properties such as tensile strength, strength-to-rupture life, and so forth, as well as thermal conductivity must determine the choice of the material. A compilation of the thermal conductivity over usable temperature ranges for various materials is given in figure 3. Additional sources are the references cited under topic designation II-A-2. A comparison of the conductivity ranges covered by several general classifications of materials is given in figure 4. In general, it may be noted that heat-resistant alloys have values of thermal conductivity ranging from approximately 7 to 20 Btu/(hr)(ft)($^{\circ}F$) over a complete temperature range. Furthermore, the conductivity is fairly constant over the entire temperature range. Plain carbon steels, however, vary from 42 to 15 Btu/(hr)(ft)($^{\circ}F$), showing decreasing values of conductivity as the temperature increases. In itself this tendency for the conductivity to decrease as the temperature increases is

undesirable from a cooling standpoint. However, the displacement between the two groups of curves is such that they intersect at temperatures near the upper limit of the temperature range over which turbine blades can safely be operated. Consequently, plain carbon steels offer higher values of thermal conductivity for the temperature range over which turbine blades are generally operated.

II-A-4. Selection of blade material - fabrication

The two methods of fabrication that should be considered are machining and casting. The machinability of the various high-temperature alloys has always been a difficult problem requiring special cutting tools such as tungsten carbide. The expense and time required for machining has led to an increasing degree of experimentation with casting procedures. The sand-casting method is limited to low melting temperature alloys because silica grains that affect metal properties are readily picked up by the metal at high temperatures. This method is therefore not adequate for high-melting-point turbine-blade materials. The "mercast" process, which employs frozen mercury, is rapidly being developed and promises to be of particular value in casting blades with cooling passages and similar intricate configurations. The "lost-wax" process of centrifugal casting has been used at the NACA for casting hollow experimental turbine blades with some success. A brief synopsis of some of the results obtained with this method follows. AMS 5385 has the best casting properties of the heat-resistant materials providing a smooth finish, although for a long length it may be difficult to make the AMS 5385 flow with a thickness less than 1/16 inch; X-40 and S816 have good casting properties approximately the same as AMS 5385. Hastelloy-B and N-155 can also be cast successfully. Although the manufacturers claim that Inconel X cannot be cast, the NACA has attempted to cast small pieces. It has been found that, during the melting process, the titanium content is reduced, thus affecting the strength properties. AISI 403 has fair casting properties. Among the lower-temperature alloys, SAE 4130 has been used to cast hollow blades; however, some difficulty has been experienced with gas porosity in the finished product. A close control of the casting process is required with 4130 and hot-tear and shrinkage difficulties were encountered. Similar difficulties are expected when attempts are made to cast Timken 17-22A alloy.

II-B-(1,2,3). Cooled-blade manufacture

Insertion of coolant passages in turbine blades can be achieved by three major methods: drilling, casting, and material disintegration.

A brief summation of the advantages, disadvantages, and pertinent facts in regard to each follows.

Drilling permits good control of hole size. Most turbine-blade materials can be drilled although special drills are required in some cases. For example, to drill AMS 5385, which is a very difficult material to machine, a carboloid drill must be employed. Difficulties are encountered with drill drift with small-diameter drills. NACA shop estimates of the approximate drill drift that will be encountered with several blade materials for various diameter holes, assuming that the proper setup (drill jig, etc.) is employed, are shown in the following table:

Material	Hole length (in.)	Hole diameter (in.)	Approximate drift (in.)
Inconel X, X-40 and similar heat-resistant alloys	3	0.060	0.030
Do ---------------------	3	.040	.060
Do ---------------------	3	.030	.120
Stainless steel and softer alloys	3	.040	.030

In casting coolant passages in blades, exact hole location is difficult. The location of holes that can be cast in a blade is limited by whether the material will flow between the tubes or cores. Excellent success has been attained with casting hollow blades of AMS 5385, but certain high-temperature materials such as Inconel X cannot be cast by present-day methods. One of the chief difficulties in casting blades with coolant passages is the problem of cores. The cores must be tapered to facilitate removal. Several coring materials have been tried at the NACA, such as ceramics, tungsten wire, and tubing of molybdenum and stainless steel. Ceramic cores are generally difficult to remove, usually requiring drilling. Tungsten-wire cores can be removed more readily by electrolytic etching. The insertion of tubing material that forms the passage walls has also been tried with some success. Frequently, a rough coolant surface results in cast blades, depending largely on the finish of the core. If the same laws that apply to stationary tubes apply in this case, such roughness is good from the standpoint of heat transfer but not from the standpoint of pressure drop required to pass the coolant through such passages.

A third method of providing coolant passages in blades is by means of the disintegrator. A carbon arc virtually burns a hole in the material. Control of the hole diameter is difficult with this method, being approximately ±0.005 inch on the diameter. A rough surface is produced and hole diameters down to a minimum of 0.020 inch can be achieved. Until recently this method was extremely time-consuming. New machines have been developed, however, which are as much as 20 times faster so that this method is now virtually as fast as drilling.

It should be noted that possible combinations of these methods might be applicable to providing coolant passages in blades.

II-B-4. Fabrication of high-conductivity inserts

NACA experience has indicated that insertion of copper strips in the blade trailing edge can be accomplished in the following manner. The blade trailing edge is slotted carefully, maintaining the slot width not more than 0.002 inch greater than the insert thickness. The tight fit is necessary in order to insure that the braze material will take hold and provide sufficient strength. The copper is next coated with flux and groundup brazing compound and then inserted into the blade slot. In the case where nickel is a major constituent of the blade material, difficulty may be experienced because embrittlement and cracking of the braze may result after firing. The best results to date were obtained with a 72-percent copper, 28-percent silver brazing compound for these applications. When the insert properly coated with flux and brazing compound has been put into the blade, the assembly is placed into a furnace with a hydrogen atmosphere and kept there for 10 to 15 minutes. The furnace temperature is maintained at 1475° F. Caution should be exercised that the original heat treatment of the blade material is not affected by this procedure. In operation the copper insert will be subjected to double shear; however, experimental runs with similar inserts placed in modified J33 air-cooled blades indicate the brazing material to be sufficiently strong to withstand the imposed centrifugal load. For the natural-convection turbine of reference 1, calculations indicate that the stress imposed by water at high rotative speeds is not prohibitive for this application. When it is added to the centrifugal stress to which the insert is subject, the result will not exceed the estimated tensile strength of the brazing material, which is 30,000 pounds per square inch ±10,000 at 600° F.

II-C-(1,2). Disk design

Stress analysis of cooled turbine disks requires use of methods that account for the thermal gradients encountered. The gradients may not be as severe as with uncooled disks, depending on whether

or not methods of cooling the outer surface of the disk are employed. The constant-strength disk method outlined in engineering handbooks does not take into account such temperature gradients. Consequently, a completely misleading picture of the actual stresses encountered could result if this method were utilized in designing a cooled turbine disk. The effect of thermal gradients upon disk stresses is considerable. Methods of stress analysis have been developed at the NACA, which include the effect of thermal gradients. These are outlined in references 15 to 17. Temperature distributions have not been obtained for the NACA water-cooled turbine disks. Furthermore the design of the two experimental NACA water-cooled turbines now being operated is such that the temperature distributions encountered are probably not indicative of results with the more conservative designs, which will probably follow.

II-D-(1,2). Coolant sealing

Two methods of coolant introduction on liquid-cooled turbines have been tried by the NACA. One consists of coolant introduction through the center of the turbine shaft and the other provides for coolant introduction by directing streams of coolant into the cavity formed by the disk and a slinger plate bolted to the disk. In both cases the water is discharged through holes drilled through an overhanging lip in the rotor at a considerable radius from the shaft center. These methods with their attendant advantages and disadvantages are fully discussed in reference 1. Sealing of the water at the blade tips was adequately achieved by screwed-in plugs and welds, which is also discussed in reference 1.

As a result of operating these turbines, it would appear that introduction of the coolant through the center of the shaft is preferable because the necessity for a rotating carbon seal is eliminated. Furthermore, when a multistage machine is considered, the number of such rotating seals required would be increased. If the design configuration is such that coolant introduction through the center of the shaft cannot be incorporated and a rotating sealing arrangement is necessary, axial rather than radial sealing with graphite seals is preferable. Operation of the 14-inch turbine (reference 1) has indicated that radial clearances of 0.001 inch are required to provide effective sealing and that the degree of effectiveness is gradually reduced as the clearance increases with turbine operation. Recent tests have shown that sealing breaks down completely after approximately 30 hours of operation. Excessive seal temperatures are not likely to occur even at high gas temperatures because sufficient lubrication is provided by the coolant and oxidation of the carbon seals does not occur until about 700° F.

These seals are manufactured and obtainable in all sizes up to a 13-inch diameter.

The method of sealing the coolant at the point of discharge on the 14-inch turbine with labyrinth seals (reference 1) was dictated by the experimental installation facilities. It is recommended that the coolant be returned to the shaft to minimize pumping losses. Adequate sealing between the rotating shaft and the stationary entrance and return coolant-supply tubes could probably be achieved by several stages of labyrinth seals. A screw-type labyrinth spiraled in the opposite direction to rotation has been found to be more effective than the circular type.

II-D-3. Sealing between blades and disk

Figure 2 shows a cross section of the rotor and the method of coolant introduction and discharge for a turbine embodying an arrangement whereby the blades are not integral with the disk but are individually removable. Each blade has a coolant reservoir in the base. Two drilled passages in the blade base connect the reservoir with the supply and return coolant passages. These are drilled radially through an adjacent disk bolted to the rotor. Coolant circulation within the blades is effected by natural convection. The inset on figure 2 shows the details of the sealing arrangement at the blade base. Copper sealing tubes are pressed between the rotor and the adjacent cooling disk. This method has not yet been tried in actual operation and there is some doubt as to the effectiveness of the sealing tubes under the combined effect of water pressure and possible differential expansion between the rotor and the adjacent cooling disk. If the coolant were introduced through radial passages drilled through the rotor into the blade base, the problem would become one of sealing between the blade base and the rotor. Lack of experimental data for the turbine shown in figure 2 prevents drawing a definite conclusion as to the effectiveness of a copper sealing tube; however, it appears to represent a good approach to the problem on the basis of knowledge currently available. Another approach would be to provide a permanent sealing method by use of brazing or welding techniques. However, the replaceability feature would be eliminated by use of such a method.

From the foregoing discussion, it is obvious that complete elimination of the sealing problem between the blades and disk is achieved by an integral construction of blades and disk. Because coolant sealing has been a major problem in the operation of water-cooled turbines, the importance of eliminating a source of possible

sealing difficulty cannot be minimized. The choice of whether turbine blades should be made individually replaceable depends, of course, upon the particular application. However, it is pertinent to dispel any misconceptions that may exist as to the economics of the problem of blade replaceability. For a small turbine blade of the size of the aluminum turbine (reference 1) and a similar solidity, it is approximately 30 percent more economical in making one turbine to machine the blades integral with one of the rotor disks. The blade height is small enough so that the cutter has sufficient rigidity and the further complication of machining fir-tree roots on the blades and matching disk is eliminated. Also the fact that the assembly procedure will be considerably complicated by any type of seal between blade and disk cannot be overlooked. Finally, present fabrication techniques do not lend themselves to providing readily a small number of replacement blades. The cost of making a small number of blades approaches that of making an entire set because of the complexity of the setting-up operations necessary in their manufacture. These facts are listed to indicate that a compromise may be required between various desirable and undesirable features in order to achieve a workable and yet economical design.

II-E-(1,2). Stator-blade cooling

Water cooling of stator blades presents several problems. One method of cooling is by means of passing the water coolant through drilled passages in the blades. The difficulties encountered with this method and suggested remedies are discussed at length in reference 1. A second method under consideration consists in passing water under pressure through a small annulus formed by the inner and outer shells of a sheet-metal blade. Each sheet-metal shell is formed to the desired blade configuration and welded along the trailing edge. Fins welded to one of the shells serve as separators when the outer shell is slid over the inner shell. The structure requires only sufficient rigidity to prevent bursting at the weld under the water pressure exerted because the stator is not a highly stressed member. Sealing problems will arise, however, in transferring the coolant from the supply source to the blades because the blades may only be firmly anchored at one end to permit expansion. Methods of eliminating such sealing difficulties are dependent to a large extent upon individual designs.

Inasmuch as the stators are not subjected to high stress loads, the need to cool them is not so imperative. In order not to suffer excessive heat loss by over-cooling, it may be advisable not to use water cooling for this purpose. The excellent advances made recently

with transpiration cooling (porous blades) might provide an adequate substitute for water-cooled stator blades. Other possibilities include the use of coated molybdenum blades that can withstand extreme temperatures if a coating sufficiently durable can be provided to prevent oxidation of the molybdenum. However, extensive use of molybdenum would probably be prohibitive because of its strategic classification. Ceramic materials should also be considered for application to stator blades. National Bureau of Standard Body 4811C represents one of the most promising of these materials from the standpoint of both heat-shock and strength properties. An investigation has been conducted at the NACA to determine the applicability of NBS Body 4811C to rotating turbine blades (references 18 and 19), which yielded promising results. In view of the sealing difficulties, possible excessive heat losses, and the design complications involved, it is recommended that other methods than water cooling be used for stator blades.

II-F-(1,2,3). Water purification

Use of water as a coolant poses the problems of material deposition and corrosion if steps are not taken to purify the coolant. Operation of water-cooled turbines at the NACA indicated that considerable material deposition resulted in the blade-cooling passages before a chemical water-purification system was installed. Operation of the Schmidt water-cooled turbine in England revealed similar material deposition due to the centrifuge action of the high-speed rotor. Also the British encountered considerable corrosion in the blade passages because the rotor steel did not have a high chromium content. In view of the desirability of using nonstrategic materials, the corrosion problem is of great importance.

The chemical water-purification system employed by the NACA has proved satisfactory. In addition, a degasifier has been added to remove the carbon dioxide present in the water, which might cause air locks in the coolant passages and so affect the heat-transfer rate. Water with an impurity content of five parts in a million results as compared to ordinary city water with impurities of approximately 150 parts per million. Obviously such an elaborate system is not practical for many applications; however, for a stationary experimental installation it is adequate. For any installation either copper tubing or galvanized piping should be included.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
Cleveland, Ohio.

REFERENCES

1. Freche, John C., and Diaguila, A. J.: Heat-Transfer and Operating Characteristics of Aluminum Forced-Convection and Stainless-Steel Natural-Convection Water-Cooled Single-Stage Turbines. NACA RM E50D03a, 1950.

2. Eckert, E. R. G., and Jackson, Thomas W.: Analytical Investigation of Flow and Heat Transfer in Coolant Passages of Free-Convection Liquid-Cooled Turbines. NACA RM E50D25, 1950.

3. Ellerbrock, Herman H., Jr., and Ziemer, Robert R.: Preliminary Analysis of Problem of Determining Experimental Performance of Air-Cooled Turbine. I - Methods for Determining Heat-Transfer Characteristics. NACA RM E50A05, 1950.

4. Hubbartt, James E.: Comparison of Outside-Surface Heat-Transfer Coefficients for Cascades of Turbine Blades. NACA RM E50C28, 1950.

5. Freche, John C., and Schum, Eugene F.: Determination of Gas-to-Blade Convection Heat-Transfer Coefficients on a Forced-Convection, Water-Cooled, Single-Stage Aluminum Turbine. NACA RM E50J23, 1950.

6. Brown, W. Byron, and Donoughe, Patrick L.: Extension of Boundary-Layer Heat-Transfer Theory to Cooled Turbine Blades. NACA RM E50F02, 1950.

7. Huppert, M. C., and MacGregor, Charles: Comparison between Predicted and Observed Performance of Gas-Turbine Stator Blade Designed for Free-Vortex Flow. NACA TN 1810, 1949.

8. McAdams, William H.: Heat Transmission. McGraw-Hill Book Co., Inc., 2d ed., 1942.

9. Brown, W. Byron, and Livingood, John N. B.: Cooling of Gas Turbines. III - Analysis of Rotor and Blade Temperatures in Liquid-Cooled Gas Turbines. NACA RM E7B11c, 1947.

10. Brown, W. Byron, and Monroe, William R.: Cooling of Gas Turbines. IV - Calculated Temperature Distribution in the Trailing Part of a Turbine Blade Using Direct Liquid Cooling. NACA RM E7B11d, 1947.

11. Brown, W. Byron, and Esgar, Jack B.: Analytical Determination of Local Surface Heat-Transfer Coefficients for Cooled Turbine Blades from Measured Metal Temperatures. NACA RM E50F09, 1950.

12. Everhart, John L., Lindlief, W. Earl, Kanegis, James, Weissler, Pearl G., and Siegel, Frieda: Mechanical Properties of Metals and Alloys. Circular C447, NBS, Dec. 1, 1943.

13. Hoyt, Samuel L.: Metals and Alloys Data Book. Reinhold Pub. Corp., 1943.

14. Anon.: Metals Handbook, 1948 Edition. Am. Soc. Metals (Cleveland), 1948.

15. Manson, S. S.: The Determination of Elastic Stresses in Gas-Turbine Disks. NACA Rep. 871, 1947. (Formerly NACA TN 1279.)

16. Millenson, M. B., and Manson, S. S.: Determination of Stresses in Gas-Turbine Disks Subjected to Plastic Flow and Creep. NACA Rep. 906, 1948. (Formerly NACA TN 1636.)

17. Manson, S. S.: Direct Method of Design and Stress Analysis of Rotating Disks with Temperature Gradients. NACA Rep. 952, 1950. (Formerly NACA TN 1957.)

18. Freche, John C., and Sheflin, Bob W.: Investigation of a Gas Turbine with National Bureau of Standards Body 4811 Ceramic Rotor Blades. NACA RM E8G20, 1948.

19. Freche, John C.: Further Investigation of Gas Turbine with National Bureau of Standards Body 4811C Ceramic Rotor Blades. NACA RM E9L07, 1950.

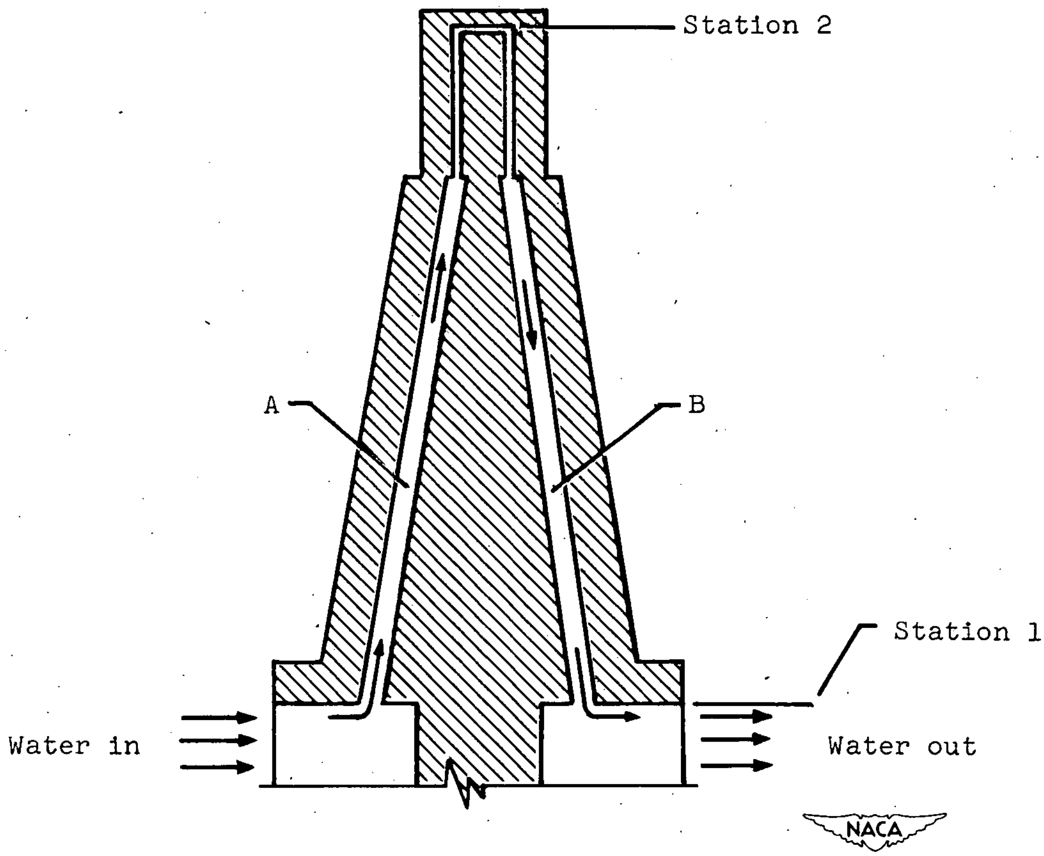

Figure 1. - Free convection provides pumping force for forced-convection cooling system.

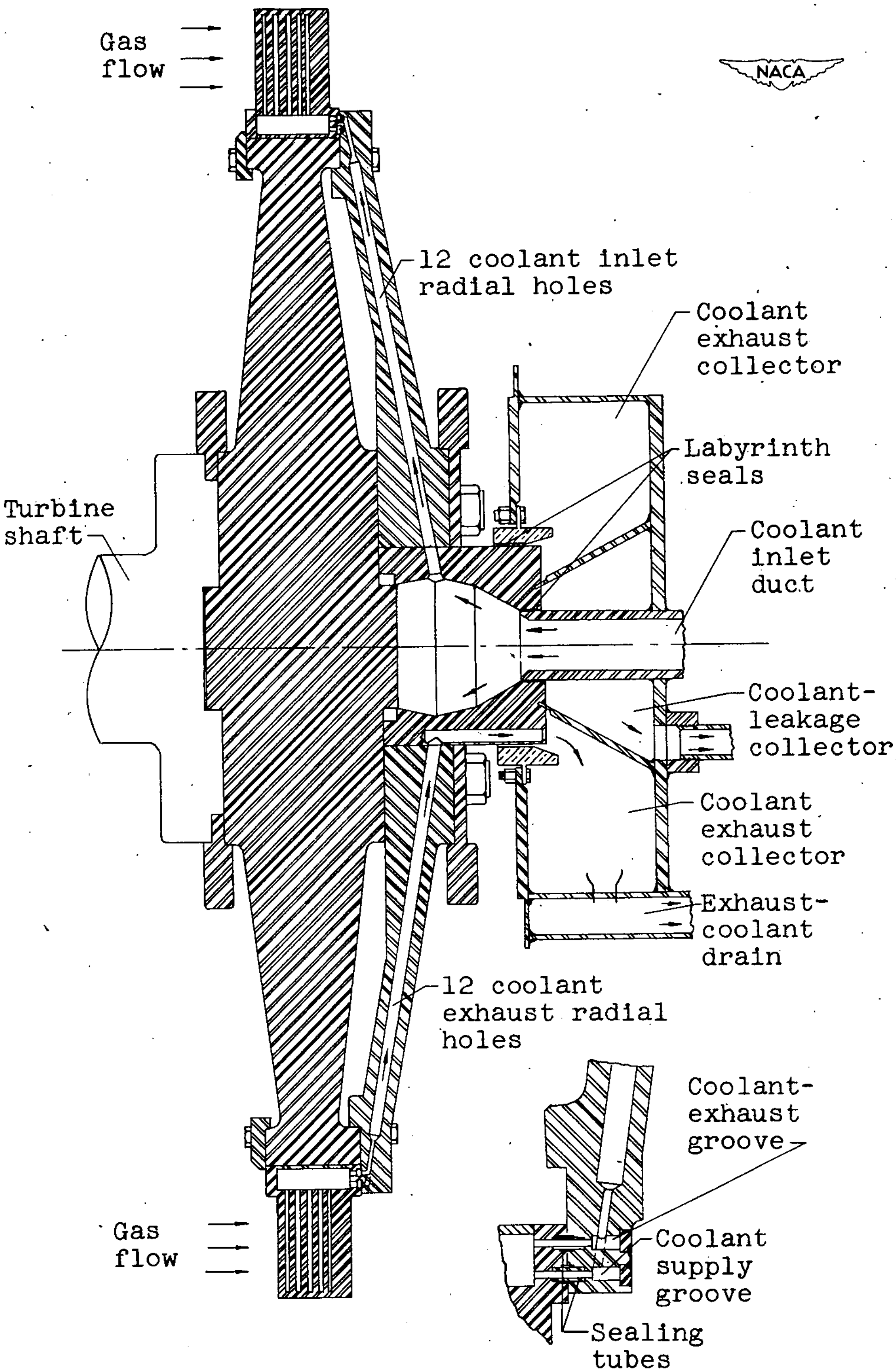

Figure 2. - Cross section of cooled turbine rotor with removable blades showing sealing arrangement.

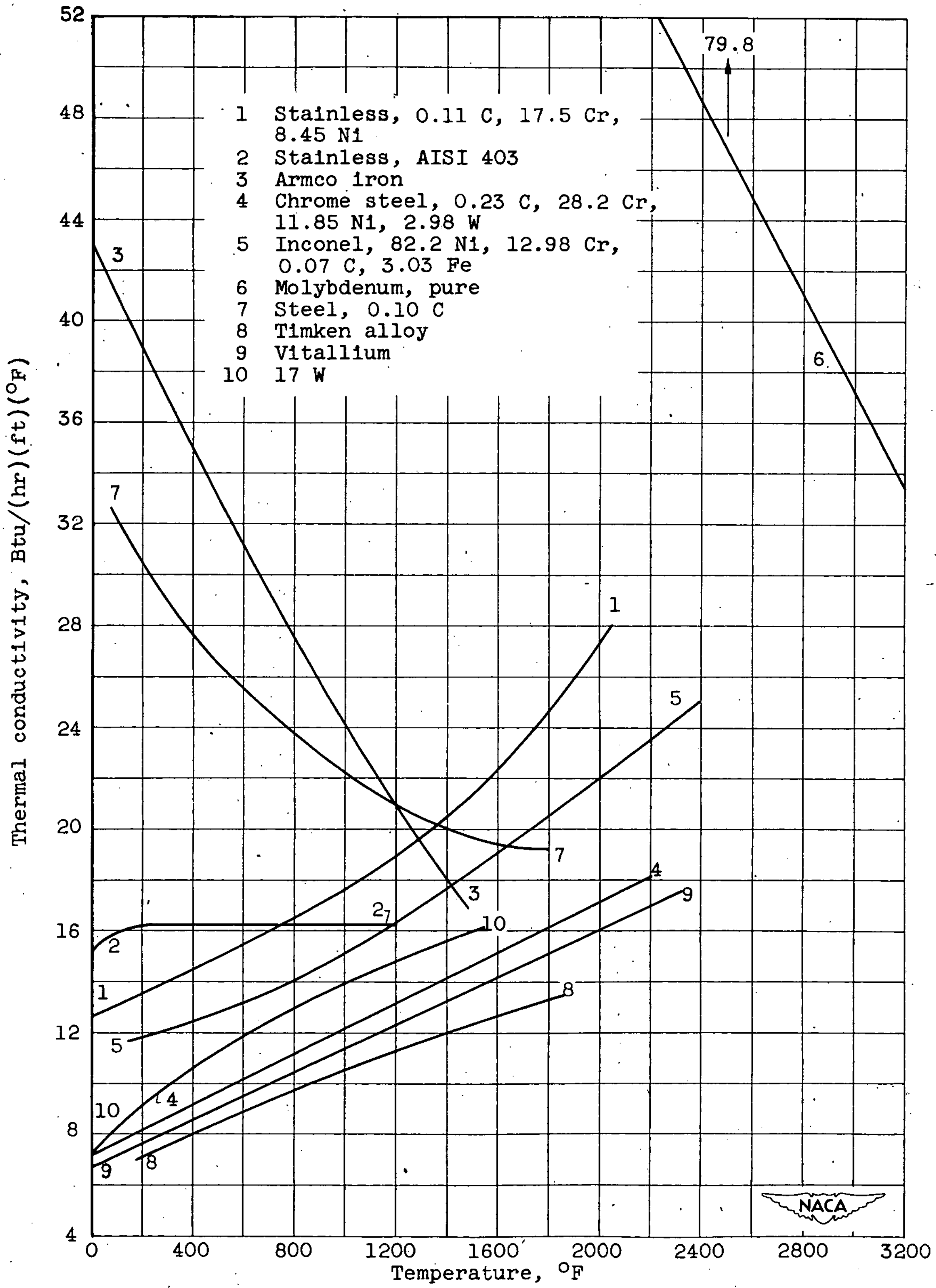

Figure 3. - Variation of thermal conductivity with temperature for various materials.

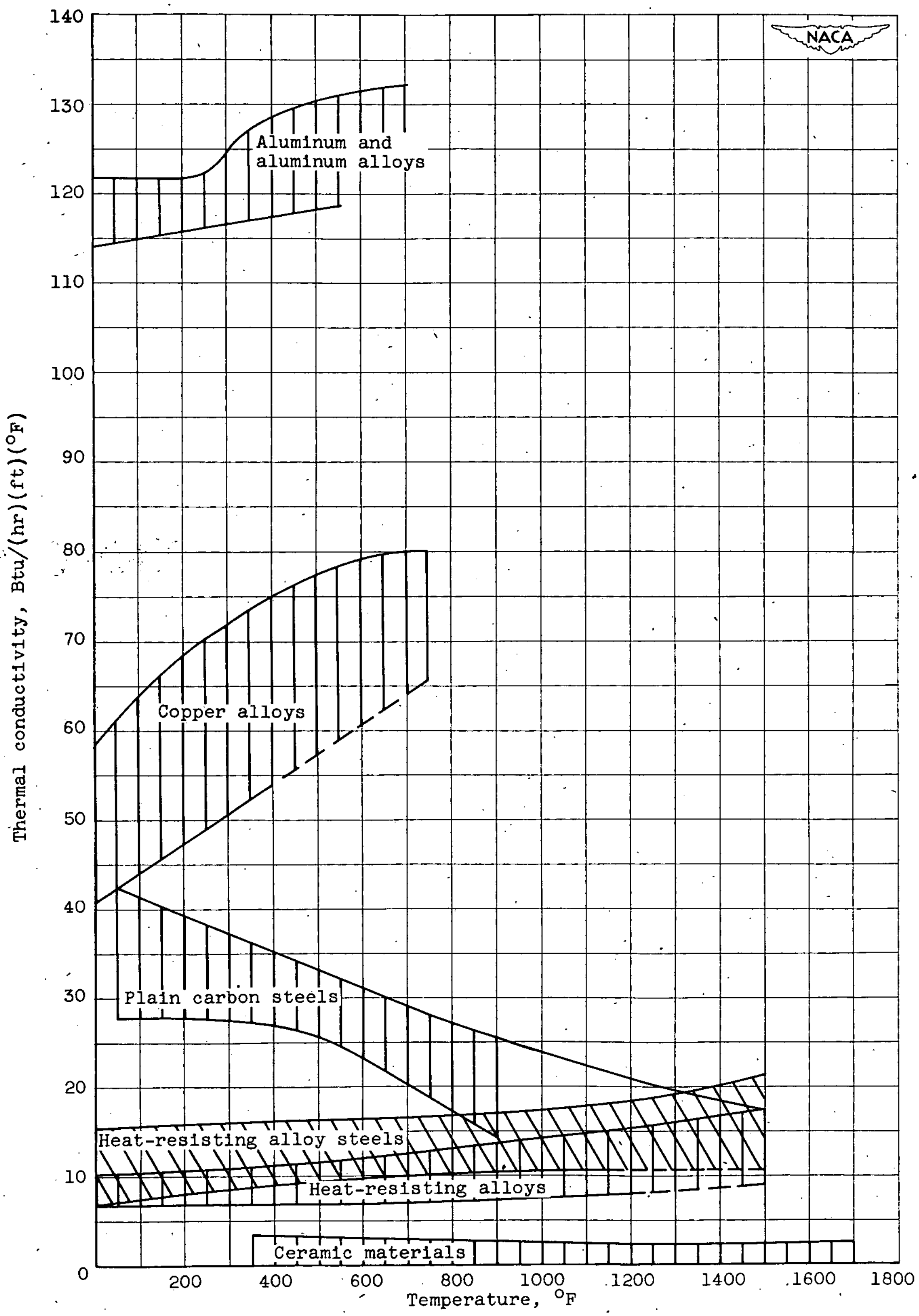

Figure 4. - Range of thermal conductivity for various materials.